Dynamic structural analysis in a 2U CubeSat considering quasi-static loads

Análisis dinámico estructural en CubeSat 2U Considerando cargas cuasi-estáticas

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ABSTRACT

Nowadays, the use of the CubeSat standard for satellites design has become a strong appliance in space industries and universities because of its low cost. In its simpler model a CubeSat , consists of a cubic nanosatellite with less than 1,33 kg and 10 cm of edge. The present work was done computational simulations using the finite element method since computational simulations are cheaper to do vibration tests. It was carried out modal analyses comparing five CubeSat structures, where it was evaluated the presence of electronic board (without electronic elements), static loads, two different aluminum alloys, and the influence of additional mass in the PCB (Printed Circuit Board). CAD software was used for the structures' design following the CubeSat specifications, and all the simulations were carried out in a finite element software, in a range of 0 to 1000 Hz. It was designed one structure to do initial simulations, followed by the design of 4 structures more adjusted to the CubeSat requirements. In the end, results showed the influences of each present condition on the satellite. It was noticed that influences of the load in the vibration shape and less than 2% in other results. The material available to build a CubeSat for tests was validated concerning the recommended aluminum alloy.

Keywords: Modal analysis, mechanical vibrations, finite elements, aluminum alloy.

RESUMEN

Hoy en día, el uso del estándar CubeSat para el diseño de satélites se ha convertido en un instrumento fuerte tanto en las industrias espaciales como en las universidades debido a su bajo costo. Un CubeSat, en su modelo más simple, consiste en un nanosatélite cúbico con menos de 1,33 kg y con 10 cm de borde. En el presente trabajo se realizaron simulaciones computacionales utilizando el método de elementos finitos ya que las simulaciones computacionales son una forma más barata de hacer pruebas de vibración. Se realizaron análisis modales comparando cinco estructuras CubeSat, donde se evaluó la presencia de placa electrónica (sin elementos electrónicos), las cargas estáticas, dos aleaciones de aluminio diferentes y la influencia de la masa adicional en el PCB (Printed Circuit Board). Se utilizó un software CAD para el diseño de estructuras siguiendo las especificaciones de CubeSat, y todas las simulaciones se llevaron a cabo en un software de elementos finitos, en un rango de 0 a 1000Hz. Se diseñó una estructura para hacer simulaciones iniciales, seguido del diseño de 4 estructuras más ajustadas a los requisitos de CubeSat. Al final se encontraron resultados que muestran las influencias de cada condición presente en el satélite. Se notaron influencias de la carga en la forma de vibración y menos del 2% en otros resultados. El material disponible para construir un CubeSat para las pruebas fue validado en relación con una aleación de aluminio recomendada.

Palabras clave: Análisis modal, vibraciones mecánicas, elementos finitos, aleación de aluminio.

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INTRODUCTION

The use of the CubeSat standard for the construction of low-scale satellites has strongly contributed to the research and practice of students and enthusiasts of the area. CubeSats are SmallSats, satellites with mass less than 180 kg. Nowadays, hundreds of universities have studied the CubeSats building, as [1] mentions. The CubeSat standard also has considerable importance in learning since it is a cheap way of satellite building and, in this way, even an unsuccessful mission leads to benefits for students in the form of experience. Not only the miniaturization but also the standardization makes easier the design of the satellite. Miniaturization and standardization make the design of the satellite easier. Also, many companies already work on manufacturing commercial kits for this model. According to [2], using a large number of small satellites allows the success of a mission due to the low cost. When many satellites are sent to a mission, and a failure occurs in one satellite, another one can replace it in the mission.

CubeSat Standard

The CubeSat standard was developed by Professor Jordi Puig-Suari, from California Polytechnic State University (Cal Poly), and Professor Bob Twiggs, from Stanford University's Space Systems Development Laboratory (SSDL), aiming to make possible the nanosatellite studies by decreasing its cost and then ensuring more access to space studies. (1) defines all the necessary CubeSat parameters so the developers can provide standard their CubeSat to be launched inside the P-POD (Poly Picosatellite Orbital Deployer). The P-POD is a structure that provides more safety to the satellite during the launching and has an ejection mechanism for when the satellite reaches orbit.

A CubeSat consists of a nanosatellite (satellite with a mass between 1 kg and 10 kg) with a cubic shape of 10 cm x 10 cm x 10 cm dimension, but it can still vary the height-based on the original cubic dimension. [1] presents all the technic specifications for dimensioning the CubeSat's structure.

Simulations with FEA in Satellites

Finite Elements Analysis (FEA) has been used in many works related to satellite modal analysis. [3] carried out modal analyses in an educational satellite, CanSat, to evaluate the structure response. They also considered static loads in the satellite. [4] presented harmonic analysis in the satellite ITUpSAT1 to find if the satellite could reach out to the PSLV launch vehicle environment using FEA. [5] performed several analyses in a Small Sat. They used modal analysis to find the modes and the effective mass, and from the values of effective mass, they sorted the principal modes to solve the harmonic analysis. With this data, they calculated the fatigue caused by vibration in their satellite by the launch vehicle Dnepr. In the end, spectrum analysis was performed to reach the random vibration effects in their structure. [6] compared the design parameter of antennas of the microsatellite UPMSat-2 to decrease the vibration in the entire structure. [7] evaluated the behavior of different parameters in honeycomb plates used in a satellite structure and compared them to experimental tests. [8] analyzed a PCB structure for a satellite and then compared it with experimental tests.

Many works also analyzed their CubeSat using FEA. [2] evaluated the stress due to a static acceleration of 10 g and the weight in a CubeSat. [9] carried out the design and analysis of a CubeSat and made analytical calculations to find the best structural dimensions considering some launcher providers' requirements. [10] carried out FEA in a designed CubeSat where they evaluated overweight in the PCB performed a structural analysis with static load, and modal analysis to verify if their CubeSat could reach the structural requirements of the launch provider. [11] evaluated FEA analysis and compared it to experimental tests in a CubeSat, MYSAT-1. The experimental tests also carried out random vibration in the satellite. [12] used FEA to perform static and modal analysis in a CubeSat, KufaSat, to compare some aluminum alloy used in satellites building. [13] compared the performance of their CubeSat built with a composite material with a commercial CubeSat made of aluminum alloy through modal and quasi-static analysis. [14] performed a static and modal analysis in their CubeSat. They also analyzed random vibration and fatigue damage analysis to verify the resistance of the satellite in the launch environment. [15], besides modal and a quasi-static analysis, coupled their analysis to heat transfer and thermal stress analysis in a CubeSat model. Other works carrying

out analysis in CubeSats similar to those cited can be found in [16-19].

Objectives

Since the vibration analysis in satellites and the CubeSat standard are very important as an affordable manner to do space studying, this work aims to carry out modal analysis in CubeSats and analyze their behavior for future experimental tests.

At first, it was designed an original structure of a 2U CubeSat (equivalent to 2 CubeSats), where it is presented in [20]. For the initial structure, this work focus on understanding the effects of static loads and the influence of a PCB on the modes. Afterward, it was designed more 4 different structures of a 2U CubeSat as improvements to the initial structure. This part of the work focuses on verifying the similarities of the CubeSats and the effect of adding mass until the maximum weight allowed by the CubeSat requirements. Also, it was analyzed comparisons of two aluminum alloys used in the building of CubeSats, one which is very used in the literature and another which is the only material available to the authors build an accurate model.

VIBRATION TESTS

[2] mentions some tests that must be done in the satellite before being launched to space to ensure it will afford all the excitation and heat generated by the rocket engine and the space environment. These specific conditions are not provided by [2] but for the launch vehicle supplier. Among these tests, there are the vibration tests.

During the launching, all the spacecraft friction forces and combustion cause significant random vibrations that can compromise the satellite structure and its electronic components. Especially when the satellite resonance frequency is found, the vibration amplitude increases and causes high fatigue failures [8]. The main vibration test objective is to verify if the satellite still is suitable to reach the mission after affording all the vehicle launching environment.

[21] presents a series of recommendations to perform vibration tests in SmallSats, mainly using an electrodynamic exciter called shaker. The main vibration test carried out in satellites are random vibration, sine-burst, and sine sweep. For small satellites, random vibration sine-burst tests are more used since they already cover the sine-sweep condition, which on the other hand, is more used for bigger satellites.

Random vibration tests are done in shakers, transmitting vibrations with random frequencies and amplitudes in a band of 20Hz to 2000 Hz. This test is more common to test the whole structure integrity, as electronic components and other devices, such as the second structure. The sine-burst test induces a sinusoidal vibration where the amplitude slightly increases, then decreases in the same frequency. This test aims to verify the principal body of the satellite, the primary structure. The sine-sweep test transmits a frequency that increases and decreases with a constant amplitude; it aims to verify the structural integrity (sine-vibe) or identify the natural frequency and damping (low-level sine sweep). According to [21], the vibration tests are also used to compare and correlate between the real model and the model acquired by finite element analysis (FEA).

MODAL ANALYSIS

Modal analysis is the process used to solve vibration equations to find the vibration modes of systems. For simple vibration analysis, where the system can be modeled as a one degree-of-freedom, the natural frequency ω and the maximum amplitude *A* are simple to be calculated. However, as the degrees-of-freedom increase, the use of computational resources such as finite element software become more suitable to solve this problem. Using FEA, one complex element is substituted by multiple minor elements. FEA makes the modal analysis of complex systems possible with acceptable approximations.

The behavior equation, free and without damping, of multiple degree-of-freedom systems, is written as equation (1):

$$[m]\vec{\ddot{x}} + [k]\vec{x} = 0 \qquad (1)$$

Where *[m]* is a mass matrix and *[k]* is a constant spring matrix, multiplying to the displacement vector's second derivative, $\vec{\ddot{x}}$, and the displacement vector, $\vec{x}$, respectively. The vector $\vec{x}$ can be replaced by $\vec{X}T(t)$, where $\vec{X}$ is a constant vector, and *T(t)* is a variable time dependent. Isolating the time-dependent terms and the matrix terms it is found equations (2) and (3):

$$\ddot{T}(t) + \omega^2 T(t) = 0 \qquad (2)$$

$$\left[\omega^2[I] - [m]^{-1}[k]\right]\overline{X} = 0 \qquad (3)$$

In equation (2), *[I]* is an identity matrix. Solving the equation (2) as an ordinary differential problem, the solution to the movement equation is found as equation (4):

$$T(t) = A\cos(\omega t + \varphi) \qquad (4)$$

Where φ is the phase angle determined by the system's initial condition. ω can only be found by the equation (3) where the non-trivial solution is equation (5):

$$\left[\omega^2[I] - [m]^{-1}[k]\right] = 0 \qquad (5)$$

And then, an eigenvalue and eigenvector problem is found. In equation (3) ω^2 is the eigenvalue and will have n different values. A matrix equation expansion to a polynomial equation will let ω^2 have n positive roots $\omega_1 < \omega_2 < \ldots < \omega_n$. For any found eigenvalue ω_i, there will be a correspondent eigenvector $\overline{X^{(i)}}$. In a general way, it is possible to write a displacement equation of a free system without damping with multiple degrees-of-freedom as equation (6):

$$\vec{x} = \sum_{i=1}^{n} \overline{X^{(i)}} A_i \cos(\omega_i t + \varphi_i) \qquad (6)$$

To verify the real displacement of a system, it would be necessary to have information about its vibration condition and orthogonalize all the eigenvectors. However, with only $\overline{X^{(i)}}$, it is possible to find the displacement relation among each element of the structure, on which it is maintained the same in a ω_i frequency applied [22]. The finite element software used in this work, Ansys Workbench 18.1, does not return real displacement nor stress values caused by the vibration since there is no excitation load to configure in the modal analysis. However, it is possible to configure the simulation specifying the maximum displacement. It is also possible to choose some orthonormalization parameters. When the orthonormalization parameters are not configured, Ansys normalize the displacement vector in each element by the equation (7):

$$\overline{X^{(i)}}^{T}[m]\overline{X^{(i)}} = 1 \qquad (7)$$

Where the term "*T*" represents the transposed function of the displacement vector and the unit term on the right side represents an identity matrix. In Ansys, this unit term depends on the configured length unit. In the simulations presented in this work, the standard normalization of the Ansys was used. Thus, the deformation and Von Mises stress results described in this work are not approximations of the real values. However, it still allows general analyses about the structural dynamic behavior. In this way, the analysis done in this work cannot supply information about the damage in the structure.

STRUCTURES UNDER STUDY

For all the CubeSat described here, it was used a CubeSat size of 2U. The CAD software used to design was SolidWorks 2016, and the design was based on the Cal Poly requirements [1].

Initial Structure
The initially, designed CubeSat height is 213 mm; however, Cal Poly in [2] defines the height as 227 mm. No ejection mechanism was designed since it is not referred to in any consulted work. The lateral plates of the structure were designed 1 mm thin. The other dimensions followed Cal Poly specifications, and additional details of the initial structure can be found in [1] and [20], respectively. The satellite CAD model made by the software SolidWorks is shown in Figure 1.

The CAD model does not have fasteners, although the structure was designed to be assembled with M3
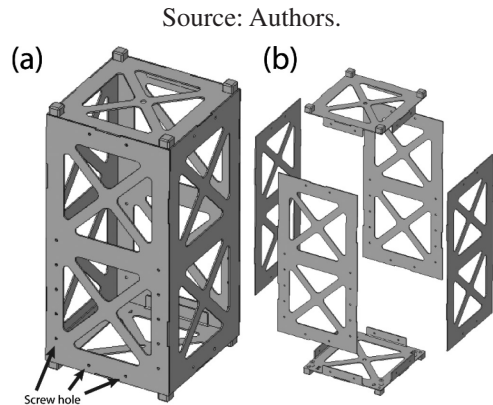
Source: Authors.

(a)   (b)

Screw hole

Figure 1. Initial designed 2U CubeSat assembled (a) and exploded (b).

screws and nuts in the lateral holes, which are shown in Figure 1. The lateral plates are bonded to the top and bottom plates by their faces to approximate the screws connections.

The principal material used in CubeSats fabrication is aluminum alloy because of its good relation between stiffness and mass, which are the principal properties to be optimized in a vibration environment. The material used in the simulation of the initial structure was a generical aluminum alloy from the Ansys materials library. Although this aluminum alloy is not referred to in [1], its principal mechanical properties (for modal analysis) are very similar to aluminum alloy 6061-T6. Aluminum alloy 6061-T6 is very used in works related to satellite design [2, 10, 12, 14] and other works as [4, 8, 9, 11] use a general aluminum alloy 6061. Table 1 shows the principal mechanical properties of the aluminum alloy of Ansys, 6061-T6, and 5083-O the material 5083-O is used in the later structures.

There are not many differences between Ansys aluminum and 6061-T6. The total structure mass is 0.19 kg, which follows the requirements from Cal Poly to be less than 2.66 kg. However, adding secondary components could increase its weight. A thin electronic plate or PCB (Printed Circuit Board) simplified with no components was added since every CubeSat has electronic components. However, the PCB considered here is just a rough approximation

of real PCBs of CubeSats since this work does not aim to evaluate a real satellite with a defined space mission. Four additional aluminum fastening brackets were also added to fix the PCB to the structure. In the simulations, the material configured to the PCB was FR-4, a polymer used in PCBs which already had its properties in the Ansys materials library. The properties of FR-4, which are anisotropic, are presented in Table 2. The material of the fastening brackets is the same as the structure. Figure 2 shows the CAD model of these components.

Because the simulation of the initial CubeSat was performed in the structure with the board and without
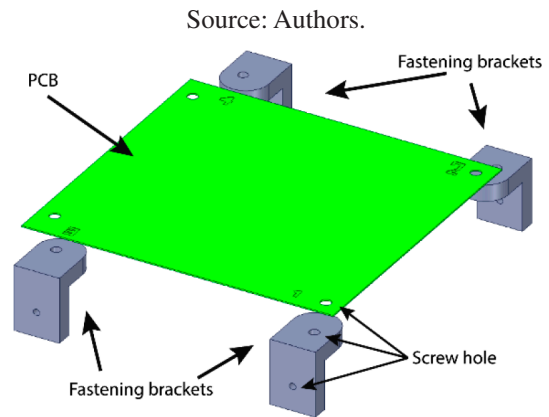
Source: Authors.

Figure 2. PCB and fastening brackets.

Table 1. Aluminum alloys properties.

Mechanical Properties	ANSYS	6061-T6*	5083-O**
Density (kg/m^3)	2770	2700	2660
Young's Modulus (GPa)	71	69	71
Shear Modulus (GPa)	26.7	26	26.4
Poisson's Ratio	0.3	0.33	0.33

Source: * www.makeitfrom.com/material-properties/6061-T6-Aluminum;
** asm.matweb.com/search/SpecificMaterial.asp?bassnum=MA5083O.

Table 2. Mechanical properties of Ansys FR-4.

Direction	All	x	y	z	xy	yz	xz
Density (kg/m^3)	1840	–	–	–	–	–	–
Young's Modulus (GPa)	–	20.4	18.4	15	–	–	–
Shear Modulus (GPa)	–	–	–	–	9.2	8.4	6.6
Poisson's Ratio	–	–	–	–	0.11	0.09	0.14

Source: Author.

the board, from now on, the structure with no PCB is referred to as NP, and the structure with PCB is denoted as WP.

Final CubeSats

After the initial CubeSat was designed, it was observed that the structure was not good enough because of weak connections, non-standardized dimensions, and inadequate thickness; thus, it was decided to improve the structure for the later construction of the satellite when 5083-O aluminum alloy plates were available. All the structure was designed to be built with those plates; small fastening brackets (like those used to fix the PCB in the initial structure) were designed to connect the plates. This structure has a height of 227 mm, and the plates are 3mm thin, as required in [1] for a 2U CubeSat. This structure can be seen in Figure 3 (a).

Another 2U CubeSat structure was designed to compare with the author's structure. This structure was based on a sketch in [2] in a 2U design and followed all the requirements from [1]. Here it is referred to as Quiroz's structure. Unlike the author's structure, the plates are designed to be connected by fasteners in the trails, which are rods with the structure's height. In both structures, it was added two empty PCBs. Quiroz's structure is shown in Figure 3 (c).

Additionally, two structures were designed. These two structures are similar to those previous two but without openings, or windows, in the plates,

as can be seen in Figure 3 (b and d). SolidWorks designed all the structures, and their connections were designed to be bonded with screws and nuts.

The principal mechanical properties of aluminum alloy 5083-O are listed in Table 1. Aluminum alloy 5083-O is usually used to build ships; however, it was the only material available to build the satellite. Table 1 shows that the principal properties of aluminum alloy 5083-O are very close to aluminum alloy 6061-T6; nevertheless, the simulations could evaluate better if this 5083-O could replace 6061-T6 in a satellite structure.

METHODOLOGY AND SIMULATIONS

Initial Structure

The mesh of the structure was left the automatically generated by the software Ansys, so the element size and method (tetrahedron elements) were kept automatic. Table 3 shows the number of nodes and mesh elements for the initial structure and the posterior designed structures.

Some simulations were carried out with Ansys, using FEA as in cited works in the literature. The structure was fixed in the foot, and the parts are bonded to each other by their face connections. As related in [20], modal frequencies were generated until the 30th mode, and just the frequencies inside the band of 0 to 1000 Hz were analyzed. This analysis verifies each mode's deformation and Von Mises
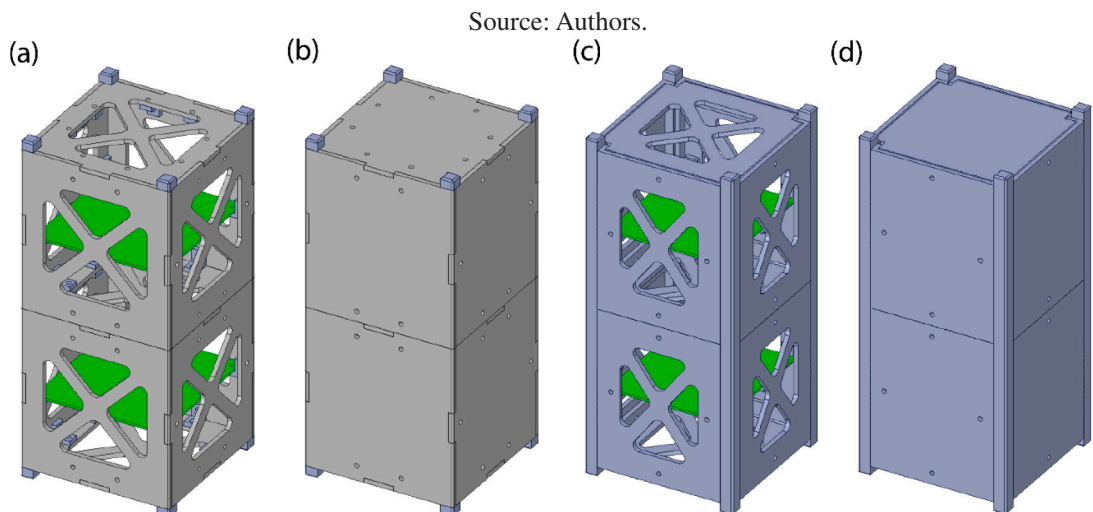
Source: Authors.

Figure 3. Designed 2U CubeSats structures: by authors, (a) and (b), and by Quiroz (c) and (d).

Table 3. Number of nodes and elements to each structure.

	Initial structure	Author's structure		Quiroz's structure	
		With windows	With no windows	With windows	With no windows
Nodes	70008	184343	206082	245952	239688
Elements	31784	44082	45898	133210	123126

Source: Authors.

stress variations. Some comparations between the structure with PCB (WP) and without PCB (NP) are performed to understand the influence of the PCB in the structure behavior. Some static loads (forces) are applied during the modal simulations to observe the variations of the results; Although, it is known that static loads should not interfere in the vibration behavior of a vibrational system. These loads are applied in different positions and directions. The value of the loads was based on [2] since there is no information about a launch vehicle to be used. The force was set with an arbitrary value of 196.2 N. This load would be the weight of a 2U CubeSat in the same environment of [2]. The configuration of loads of 5 different simulations was done as follows, following the numeration in Figure 4:

A) Load in 1 lateral plate (it can be any of the 4 lateral plates because of the symmetry) – Arrow 2, 3, 5, or 6;
B) Load in two opposite lateral plates (it can be any of the 2 opposite lateral plates because of the symmetry) – Arrows 2 and 5, or 3 and 6;
C) Load at base – Arrow 4;
D) Load at top – Arrow 1;
E) Load at both base and top – Arrow 1 and 4.

Final CubeSats
Unlike the initial CubeSat, it was decided to refine the mesh to improve the results. Different methods (hexahedral, multizone, and sweep) were used depending on the parts and from 2 mm to 3 mm element sizes. Figure 5 shows the mesh quality of all the designed structures. As it is possible to see, the quality of the final CubeSats mesh have a much superior quality than the initial CubeSat mesh. The information about nodes and the number of elements are shown in Table 3. So, the later structures have a mesh with more quality than the initial structure; however, the computational cost has increased.

The structures were fixed by the foot, the parts are bonded by their face connections to the fastening brackets. These simulations were generated and analyzed the frequency modes until right before 1000 Hz. No comparations with the presence of a PCB are made in these simulations since the results with the initial structure are enough to evaluate its influence, although all the structures have PCBs. But the presence of loads is considered again. However, in these simulations, the loads are accelerations of 1 g instead of vectorial forces and only applied in the vertical direction. Each structure performed simulations with aluminum alloy 5083-O and 6061-T6 to compare the materials. Some simulations were done considering the maximum weight in the PCB; in other words, since the CubeSat has a limit weight of 2.66kg, all the weight needed to reach 2.66kg were distributed in the two PCBs in all four
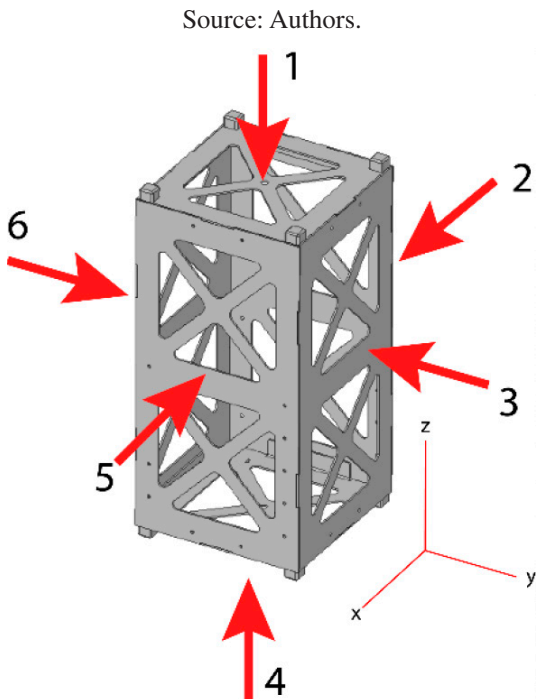
Source: Authors.

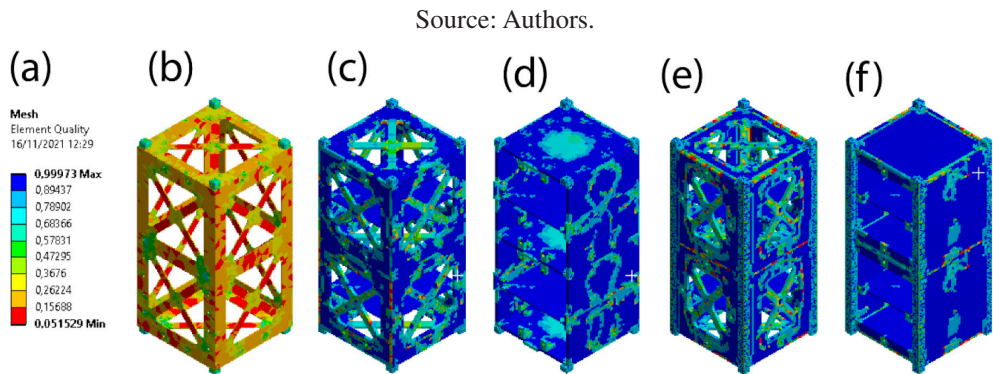Figure 4. Positions and direction of loads.

Source: Authors.

Figure 5. Mesh quality: (a) scale; (b) initial CubeSat; (c) authors' CubeSat without windows; (d) Quiroz' CubeSat; (e) Quiroz' CubeSat without windows.

Table 4. Added mass for each PCB in each CubeSat.

	5083-O		6061-T6	
	Total mass (kg)	Added mass (kg)	Total mass (kg)	Added mass (kg)
Authors with windows	0.6694	0.9953	0.6784	0.9908
Authors without windows	0.9724	0.8438	0.9864	0.8368
Quiroz with windows	1.0028	0.8286	1.0178	0.8211
Quiroz without windows	1.2288	0.7156	1.2468	0.7066

Source: Authors.

structures. Table 4 lists the mass distributed in each of the two PCBs for each structure and material for the last simulations.

RESULTS AND DISCUSSION

Initial Structure

[20] verified some additional results, but some of them are irrelevant or inaccurate. So here is presented only the principal results carried out for the initial structure.

The weight of the structure is 0.22 kg, then to find a load of 196.2 N during a launching into space, an acceleration of around 90.9 g would be necessary. This is a major acceleration, and there is no such high acceleration in any consulted work or spacecraft manual, so it is a load far above any load in a launch vehicle supplier. However, a significant mass structure could find a load of this order with an acceleration much smaller; all this additional mass could be easily found with the secondary structure. At first, it was chosen to use a vectorial force as the static load; however, the necessary acceleration to reach this force would have to be very big.

WP and NP structures were simulated to generate up to 30 mode shapes. Most of the loads do not make big changes in the results; nevertheless, few errors were observed, with a maximum of 1% in the natural frequencies. Only load A (load in 1 lateral plate) causes changes in the mode shapes of the structures, and for the other loads, most of the deformation modes stay the same.

Figure 6 shows the changes in the structure's natural frequency with a PCB.

The presence of the PCB changes many modes, and fewer have the same or a close natural frequency. In Figure 6, in most of the modes, the structure WP finds out a frequency bigger than the NP structure. The first mode appears at 302 Hz for NP structure, and for WP structure, the first mode appears only at 366 Hz.

A significant change in the mode shapes was observed with the addition of a PCB. In most mode shapes generated, the deformation is controlled by the plates of the structure. This happens even in
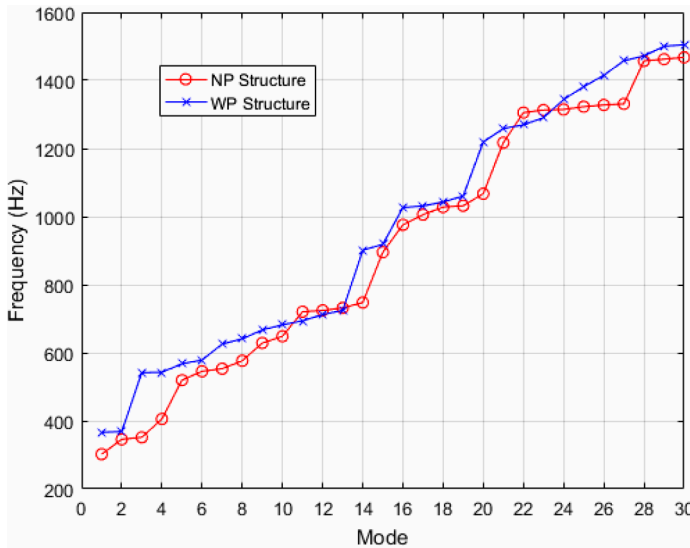
Source: Authors.

Figure 6. Comparation of natural frequencies NP and WP structures.

the WP structure since the plates are very thin and flexible. In a band of 0 to 500 Hz, the deformation concentrates in the middle of the lateral plates. In a band of 500 to 750 Hz, the deformation concentrates mostly in the windows' center, where the triangular openings are. Up to 750 Hz, the mode shapes are more arbitrary and difficult characterization.

Figure 7 shows the maximum deformation of both NP and WP structures in the band of 300 to 1000 Hz. It is not possible to compare the values between the structures of each mode because of the orthonormalization. Although in this graph, the maximum deformation peak is around 700 Hz (11th mode) in the WP structure can be visualized. This deformation is controlled by PCB,
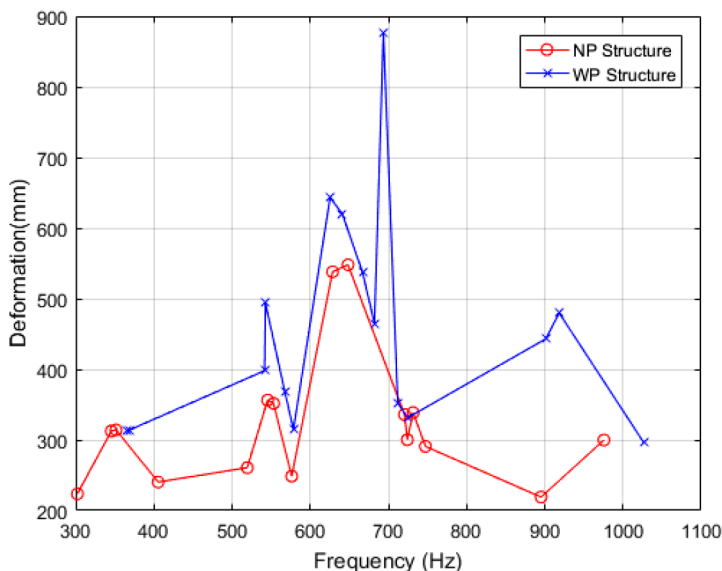
Source: Author.

Figure 7. Deformation of NP and WP structure.

where all the energy is concentrated on its center, as illustrated by Figure 8.

Final CubeSats

All the mode shapes were compared to find the influence of the openings (windows), materials, and acceleration loads. For all structures, it was observed that in the band of 0 to 1000 Hz, most of the mode shapes have the maximum deformation in the PCBs, and the maximum stress is frequently found in the fastening brackets and PCBs connections. Concerning

the mode shapes when using the two materials, at least in the band of 0 to 1000 Hz, only minor differences between the materials were found. Table 5 shows the natural frequencies of each structure using the two aluminum alloys up to the nearest mode after 1000 Hz. It is possible to see that the number of modes in both structures without windows is fewer than in the structure with windows.

Figure 9 shows the curves of natural frequencies of the 4 structures. All the structures have the first
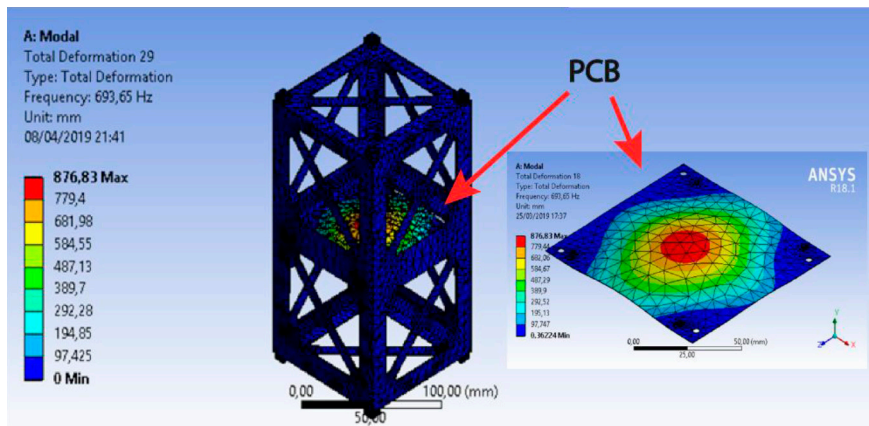
Source: Author.

Figure 8.   Deformation of WP structure FEM model on 11th mode.

Table 5. Natural frequencies (Hz) of structures with each aluminum alloy.

Mode	Author's with windows			Author's without windows		
	5083-O	6160-T6	Difference	5083-O	6160-T6	Difference
1	597.39	597.63	0.24	625.75	625.90	0.15
2	598.03	598.27	0.24	626.21	626.36	0.15
3	869.95	870.54	0.59	880.35	880.59	0.24
4	872.22	872.46	0.24	881.17	881.37	0.20
5	900.36	906.89	6.53	1002.60	1009.80	7.20
6	901.51	907.75	6.24			
7	1389.00	1389.20	0.20			
Mode	Quiroz's with windows			Quiroz's without windows		
	5083-O	6160-T6	Difference	5083-O	6160-T6	Difference
1	631.90	632.30	0.40	633.51	633.89	0.38
2	632.00	632.40	0.40	633.72	634.10	0.38
3	951.25	958.21	6.96	1001.40	1002.00	0.60
4	952.55	959.48	6.93			
5	1000.50	1001.00	0.50			

Source: Authors.
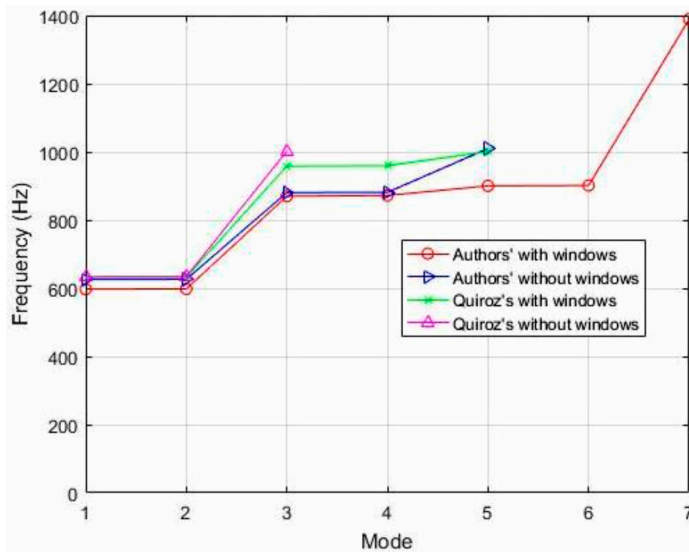
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Source: Authors.

Figure 9.   Natural frequency for each structure.

mode at about 600 Hz, and the author's structure without windows shows natural frequencies slightly smaller than the others.

The load causes a maximum error of 2% concerning the results without loads in all the results. These simulations did not find any big differences in the mode shapes like in the simulations of the initial CubeSat design.

The simulations performed with a critical mass in the PCBs increase considerably the number of modes, as shown in Figure 10. The number of modes goes from less than 10 modes to about 40 modes. The decreasing of modes without the windows still happens in these conditions. The increase of mass on the PCBs decreases the natural frequency, as stated by equation (5). Since the mass added to each structure is larger than the original structure mass, the natural
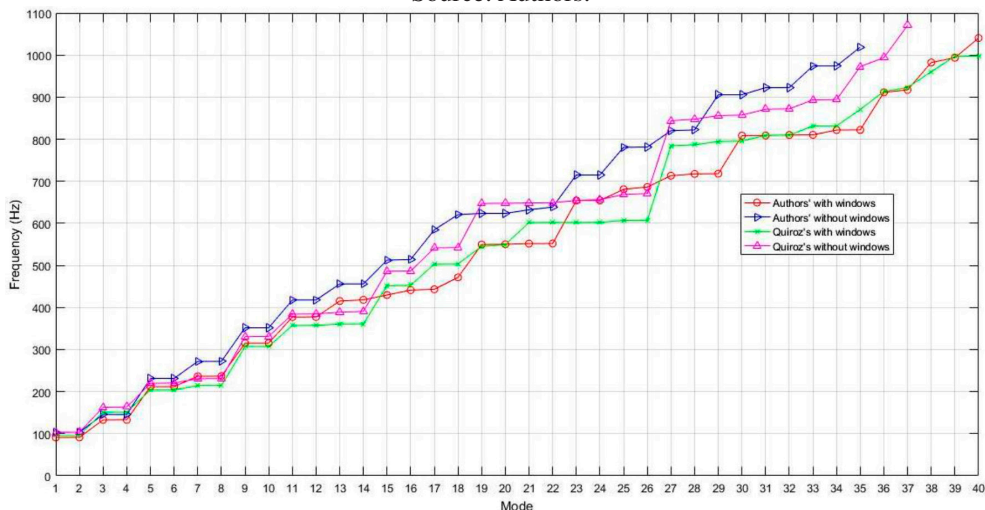
Source: Authors.

Figure 10. Natural frequencies for each structure with critical mass in PCBs.

frequencies decrease proportionally, and therefore more modes get in the range of 0 to 1000Hz.

Almost all the observations done to the structure without critical mass in the PCBs are the same as structures with critical mass regarding the two materials. However, Quiroz's structure without windows has a different behavior, as shown in Figure 11 and Figure 12.

The Quiroz's structure without openings behaves differently with each aluminum alloy used when there
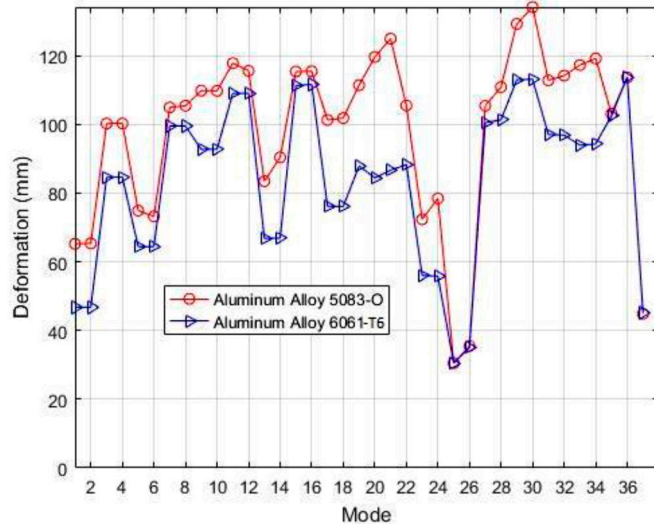
Source: Authors.

Figure 11. Deformation for two materials in Quiroz's structure with no openings and critical mass.
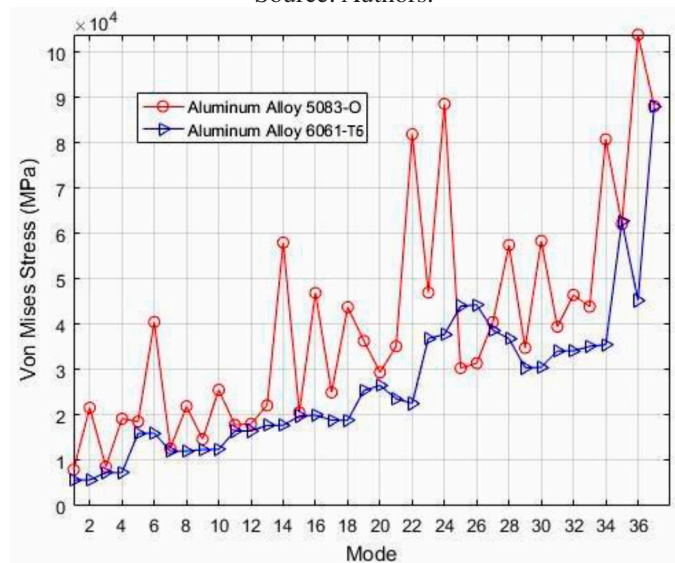
Source: Authors.

Figure 12. Deformation for two materials in Quiroz's structure with no openings and critical mass.

is a critical mass, although, both materials behave the same in all other simulations. This behavior happens only with this structure since the other three have de same behavior using both materials. As the graphs show, in almost all the modes, the aluminum alloy 5083-O has cosiderable deformation and stress than aluminum alloy 6061-T6.

It is possible to see in Figure 11 that there is a high reduction of deformation on modes 25 and 26; those modes are controlled by the whole structure, not only the PCB. Therefore, the deformation is lower since the satellite structure is more rigid than the PCB. Besides that, the maximum stress is concentrated at the foot in these modes.

CONCLUSIONS

This work presented two sets of simulations carried out in CubeSats models. In the first part, the initial CubeSat was designed, and it was performed series of simulations as it is also related in [20]. The initial structure was very light and flexible, and other secondary structures could change the results drastically since even a thin PCB without components changed the mode shapes in a band of 1kHz. Most of the quasi-static loads applied did not affect the results. However, load A (load in 1 lateral plate) affected the mode shapes generated. This load could be understood as a negative influence on the accuracy of the modal analysis using FEA when static loads are present since static loads should not affect modal features. Because of this, a modal analysis was carried out to see the behavior in loads represented as acceleration in the second part.

In the second set of simulations, now in four different structures, there were found influences of the structure's openings, which decreased modes of vibration in the range of 1000 Hz because of the decreasing of the mass and rigidity of the structure. It was also found that all the structures designed have the first mode at about 600 Hz. The reason for this similarity of the 4 structures could be the thickness (3 mm), which is the same. Therefore, these structures could be used and adapted for CubeSat developers with the certainty of having a high first natural frequency. Also, most of the modes are controlled by the PCBs, since they are the less rigid part of the structure.

The modal analysis with static acceleration loads showed that vectorial load could modify the results, but at most 2% of error, in the initial structure.

In general, there were no differences between the results when using 5083-O and 6061-T6 aluminum alloy for the four structures. However, this was not completely true with the addition of mass on the PCBs. In the case of Quiroz's structure with no windows, different results of deformation and Von Misses stresses where found when using each of the two materials. Since the other structures did not present such behavior, this behavior may be a problem of the FEA analysis carried out.

The saved data can be used to compare with real vibration tests. Also, the material available to build a CubeSat was validated since most of the results were very similar to the results for a material often used in satellites design. In this way, the future experimental test will be carried out to validate the found results and CubeSat designers will be able to use these results to improve their satellite analyses.

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